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MECHANICAL ENGINEERING DEPARTMENT

SCHOOL OF ENGINEERING

THE UNIVERSITY OF CONNECTICUT

INTERIM PROGRESS REPORT for the Period November 1, 1963 through April 30, 1964

on

HEAT TRANSFER STUDIES OF VAPOR CONDENSING AT HIGH VELOCITIES IN STRAIGHT TUBES

Reported by

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Mechanical Engineering Department
University of Connecticut
Storrs, Connecticut

May 1, 1964

Dr. T. L. K. Smull, Director
Grants and Research Contracts
NASA
Washington 25, D. C.

Dear Sir:

The following interim report covers the research done under Contract No. NASA Code SC-NsG-204-62 for the period November 1, 1963 through April 30, 1964.

This work is a continuation of that performed under the same contract number for the period November 1, 1961 through October 31, 1963. In that contract period a series of non-linear, first order differential equations was derived to attempt to mathematically describe the high speed flow of a condensing vapor. Reference 1 of this report describes in detail these equations as well as their solution on a digital computer. The flow conditions used in the solution of the equations were based upon data for steam condensing in long, small diameter tubes, obtained from a series of tests (Refs. 2 and 3 of this report) performed at the University of Connecticut in the early 1950's. For several different entrance conditions and heat transfer rates it was possible to obtain a satisfactory correlation of the condensing heat transfer coefficients with the flow parameters. It was not possible, however, to obtain a suitable correlation of the two-phase friction factor.

It was decided that the difficulty lay in the interpretation of some of the test data, particularly the total pressure (vapor velocity) measurements. The first objective, therefore, of the present research is to obtain a more accurate set of data; secondly, to further develope the analytical treatment of the annular flow model.

This report covers the initial phase of this experimental work and describes the new test apparatus and changes in the analytical procedures.

Yours truly,

Charles H. Coogan Jr

Charles H. Coogan, Jr.
Professor and Head of
Mechanical Engineering Department

INDEX

Description of Test Apparatus

The test apparatus has been designed with a different objective than the earlier equipment which was intended primarily to measure heat transfer rates in condensing flow. It is arranged primarily to determine pressure drop as a function of the heat transfer rate.

The test rig consists of a straight, concentric tube heat exchanger; the inner tube, twenty feet long with a 1.055 inch inside diameter and 0.035 inch wall, carrying the steam and the outer tube, nineteen feet long with a 2.009 inch inside diameter and a 0.058 inch wall, forming the cooling water annulus. Both tubes are commercial hard drawn copper. Figs. 1 and 2 show the overall layout and a typical cross section of the test apparatus. By reversing the direction of the cooling water flow, the test section can be operated in either a parallel or counter flow manner.

To obtain the desired measurements, the test section is instrumented in the following manner:

i) Wall static pressure taps on the inner steam pipe formed by No. 60 drill holes connected to the outside manometers through hypodermic needle tubing.

ii) Copper-constantan thermocouples to determine the temperature on the outer wall of the inner tube.

iii) Copper-constantan thermocouples in the cooling water annulus.

iv) An axially traversing total pressure probe of the Dussourd type. (Ref. 4.)

Table I shows the axial location of these measurement points.

At the tube entrance are located a static pressure tap, and radially traversing total pressure and total temperature probes. From these

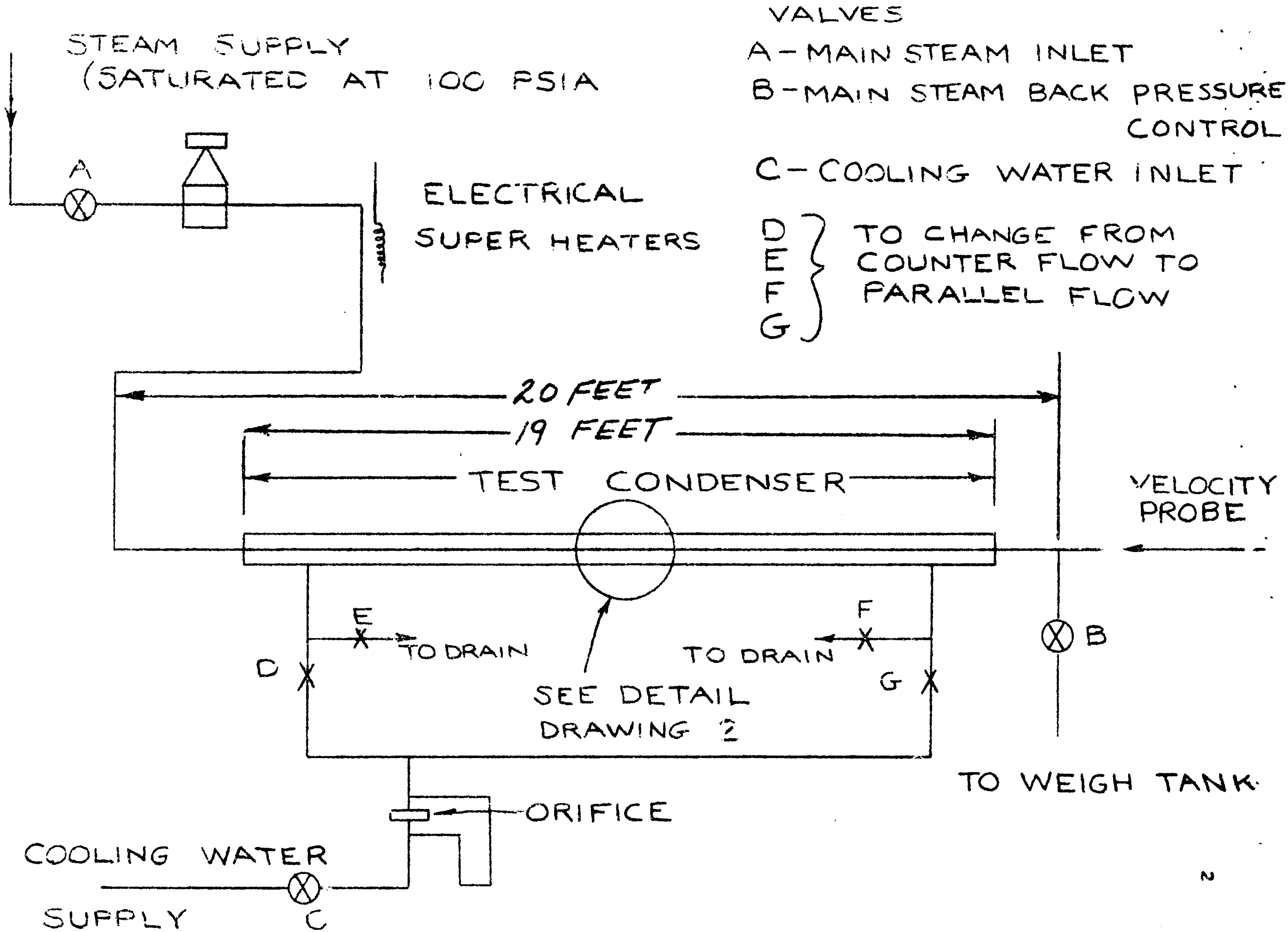
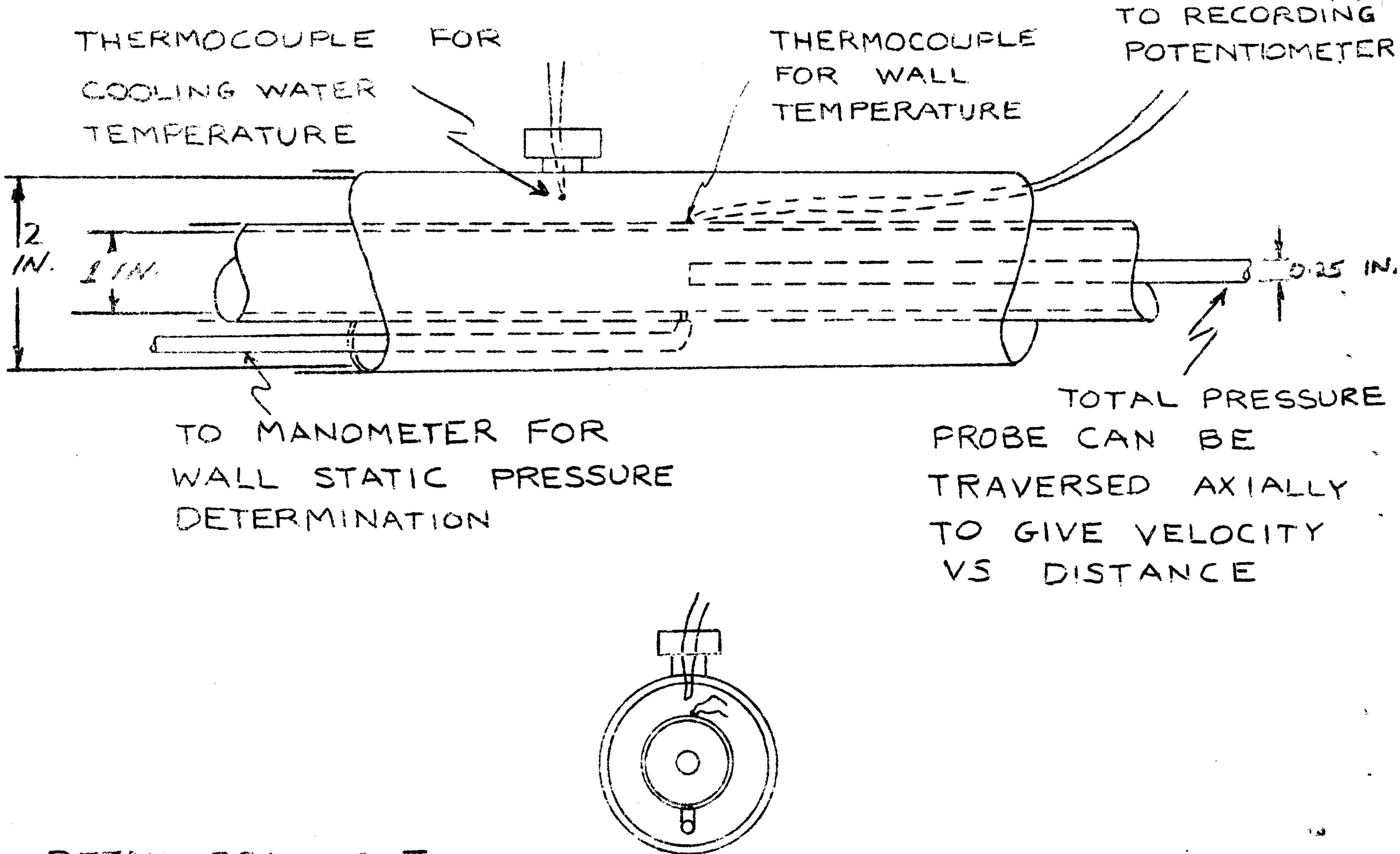

STEAM SUPPLY
(SATURATED AT 100 PSIA
A
ELECTRICAL
SUPER HEATERS
VALVES
A-MAIN STEAM INLET
B-MAIN STEAM BACK PRESSURE CONTROL
C-COOLING WATER INLET
D
E
F
G
TO CHANGE FROM COUNTER FLOW TO PARALLEL FLOW
20 FEET
19 FEET
TEST CONDENSER
VELOCITY PROBE
E
TO DRAIN
F
TO DRAIN
D
G
B
SEE DETAIL DRAWING 2
TO WEIGH TANK
ORIFICE
COOLING WATER
SUPPLY
C

FIGURE 1



DETAIL DRAWING II

FIGURE 2

Table I

<u>Measurement Points</u>

P - Indicates provision to make static pressure measurement

TC - Indicates provision to make temperature measurement

Position Number	Axial Position Ft. from Inlet	Wall of Inner Tube	Outer Wall of Inner Tube	Annulus Water Thermocouples
1				
2	1	P	TC	
3	1½	P	TC	TC
4	2	P	TC	
5	2½	P	TC	TC
6	3	P	TC	
7	3½			TC
8	4	P	TC	
9	4½			TC
10	5	P	TC	
11	5½			TC
12	6			
13	6½			TC
14	7	P	TC	
15	7½			TC
16	8			
17	8½			TC
18	9	P	TC	
19	9½			
20	10			TC
21	10½			
22	11	P	TC	
23	11½			TC
24	12			
25	12½			TC
26	13	P	TC	
27	13½			TC
28	14	P	TC	
29	14½			TC
30	15	P	TC	
31	15½	P	TC	TC
32	16	P	TC	
33	16½	P	TC	TC
34	17			
35	17½	P	TC	TC
36	18			
37	18½	P	TC	TC
38	19			

At position number 1, provision is made to measure total and static pressure and to measure the total temperature of the flowing steam. Position 1 is located 3 inches from the inlet of the tube.

measurements, the flow rate of the steam is computed and compared to that determined by weighing the condensate.

The total pressure probe of item iv is shown in Fig. 3. Briefly, it consists of a conventional total tube with the addition of static taps in the wall of the tube. These taps are located at distances from the tube mouth of 0.1, 0.2, 0.4, and 1.0 inches. When vapor with entrained liquid enters the mouth of the tube the vapor is quickly brought to rest, but the liquid, with its greater momentum, continues down the tube. A standard total pressure tube would read the combination of the liquid and vapor momentum change, giving an erroneous total pressure for the vapor alone. With the Dussourd-type probe, however, the static tap at 0.1 inch sees very nearly the true vapor stagnation pressure while the tap at 1.0 inch will, by manipulating the liquid interface in the total tube, see the combined pressure. The taps at 0.2 inches and 0.4 inches will give intermediate pressures. By plotting these pressures versus their axial location on the probe, the true vapor total pressure at the tube entrance can be obtained by extrapolation. This total pressure will be used to determine the steam velocity.

Test Facility Capabilities

As presently constructed, the test apparatus has the following maximum capabilities:

a.)	cooling water flow rate	18,000 lbm/hr
b.)	regulated steam pressure	55 psia
c.)	steam flow rate	1500 lbm/hr
d.)	installed super heaters	25 k.w.

The experimental data can be obtained to within the following accuracies:

Steam flow rate	± 2%	Cooling water temperature	± 1%
Water flow rate	± 2%	Static pressure	± 2%
Tube wall temperature	± 4%	Dynamic pressure	± 2%

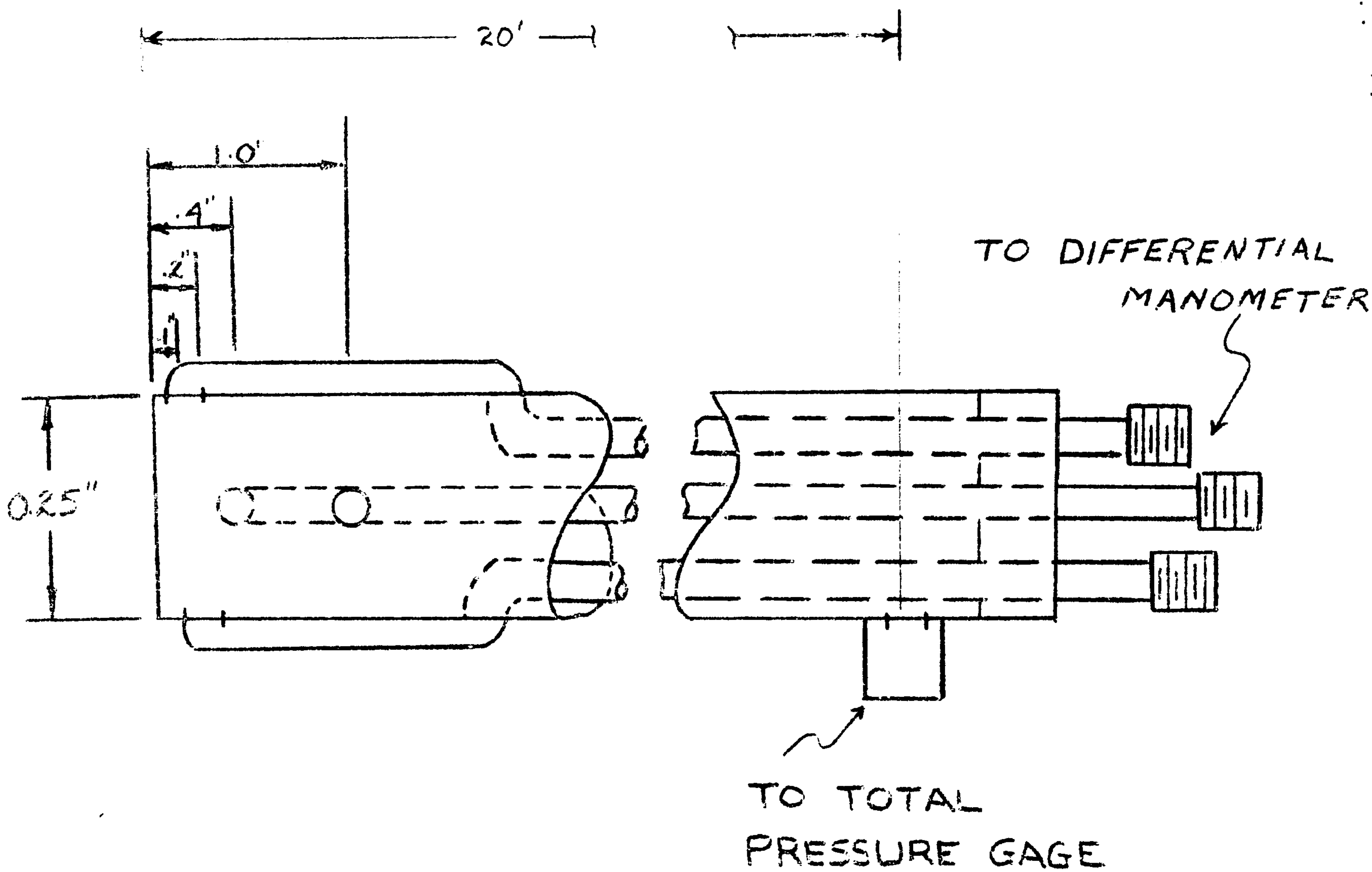


FIGURE 3
TOTAL PRESSURE PROBE

Procedure

Briefly, the test procedure is as follows: The manometers and lines filled with water and balanced; cooling water flow pattern determined and flow initiated; steam flow turned on and equilibrium flow rate established; velocity probe inserted to most forward position and then withdrawn, readings of total pressure made at prescribed axial positions; static pressure readings made with probe in and probe out.

The data obtained from the test apparatus is used as input for a data reduction program for the IBM 7040 computer. The information will be inserted in the form as recorded from the tests; i.e., pressures in psig and inches of mercury, temperatures in $^{\circ}F$ and flow rates in lbm/hr. This input will then be converted to consistent units and the data processed to produce as output the static and dynamic pressures in psia, tube wall and cooling water temperatures in $^{\circ}F$, vapor and liquid velocities, the physical properties of saturated steam and saturated liquid such as enthalpy, density and entropy at each measuring point, and the condensing heat transfer coefficient in Btu/hr ft^2 $^{\circ}F$. Tables II and III illustrate the input and output form.

Steam Table Computer Program

Appendix A of Ref. 1 describes a method of predicting steam properties from entrance conditions and local properties. However, this method does not give the degree of accuracy now desired so a method making use of a computer sub-routine (Refs. 5 and 6) for the steam table properties was developed. This sub-routine determines enthalpy, entropy and specific volume as a function of pressure and temperature in the superheat region; in the saturated region, the vapor enthalpy, entropy and specific volume are found as functions of pressure; the saturated liquid enthalpy, entropy and specific volume are given

Table II

Input Data

Data Reduction Program

NASA Project

Mechanical Engineering Department

Stagnation Chamber

Pressure (psig)

Temperature (°F)

Cooling Water Conditions

Flow Rate (lbm/hr)

Type Flow

Distance	Static Pressure		Stag. Pressure		Outside Wall Temp.	Cooling Water Temp.
(ft)	(in Hg)	(psia)	(in Hg)	(psia)	(°F)	(°F)

Table III

<u>Output Data</u>

Data Reduction Program

NASA Project

Mechanical Engineering Department

Stagnation Chamber	Cooling Water Conditions
Pressure (psia)	Flow Rate (lbm/hr)
Temperature (^oF)	Type Flow

Distance				Saturated Vapor Properties				
Actual	Dimensionless	Stag. Pres.	Stat. Pres.	Temp.	Density	Velocity	Flow Rate	Area
(ft)		(psia)	(psia)	(^oF)	(lbm/ft^3)	(ft/sec)	(lbm/hr)	(ft^2)

Distance				Saturated Liquid Properties				
Actual	Dimensionless	Stag. Pres.	Stat. Pres.	Temp.	Density	Velocity	Flow Rate	Area
(ft)		(psia)	(psia)	(^oF)	(lbm/ft^3)	(ft/sec)	(lbm/hr)	(ft^2)

Distance				Cooling Water Properties		
Actual	Dimensionless	Temp.	Heat Rate	Outside Wall Temp.	Inside Wall Temp.	Surface Coefficient
(ft)		(^oF)	(Btu/hr)	(^oF)	(^oF)	($Btu/hr - ft^2\ ^oF$)

as functions of temperature and temperature and pressure are found as functions of each other on the saturation line.

A second sub-routine has been developed that calculates the properties of steam for a constant entropy process, using the steam table sub-routine. Valid results are obtained in the wet, saturated, and superheated regions. Given the properties at one point and the pressure at a second point, the routine will calculate the remaining properties at both points assuming an isentropic process.

Projected Work

During the next period, the test work previously described will be continued. The data processing and the computer program for the solution of the flow equations will be under continual revision. Using data from the existing test section, a method involving the use of successively smaller finite difference equations will be used to calculate a local wall friction factor. This computation is based on the momentum equation and requires an accurate knowledge of the vapor velocity.

The problem of non-condensables in the steam is being investigated and some experimental apparatus obtained to gain quantitative information of air entrainment. This is a quite difficult problem and as of this date, no quantitative data has been obtained. Further work in this area is contemplated.

Additional Apparatus to be Constructed

Additional experimental apparatus will be constructed. It is expected that an annular heat exchanger of the same radial dimensions as the existing test test section but approximately two feet long will be built. This section will be instrumented with electrical resistance probes to measure liquid layer thickness, velocity probes to traverse the vapor flow in the radial direction and liquid layer velocity probes. The cooling water jacket will be closely monitored so that accurate heat transfer rates can be determined. The above data will be used in an attempt to develope a successful correlation of the local wall friction factor with the flow parameters and liquid layer thickness for condensing steam.

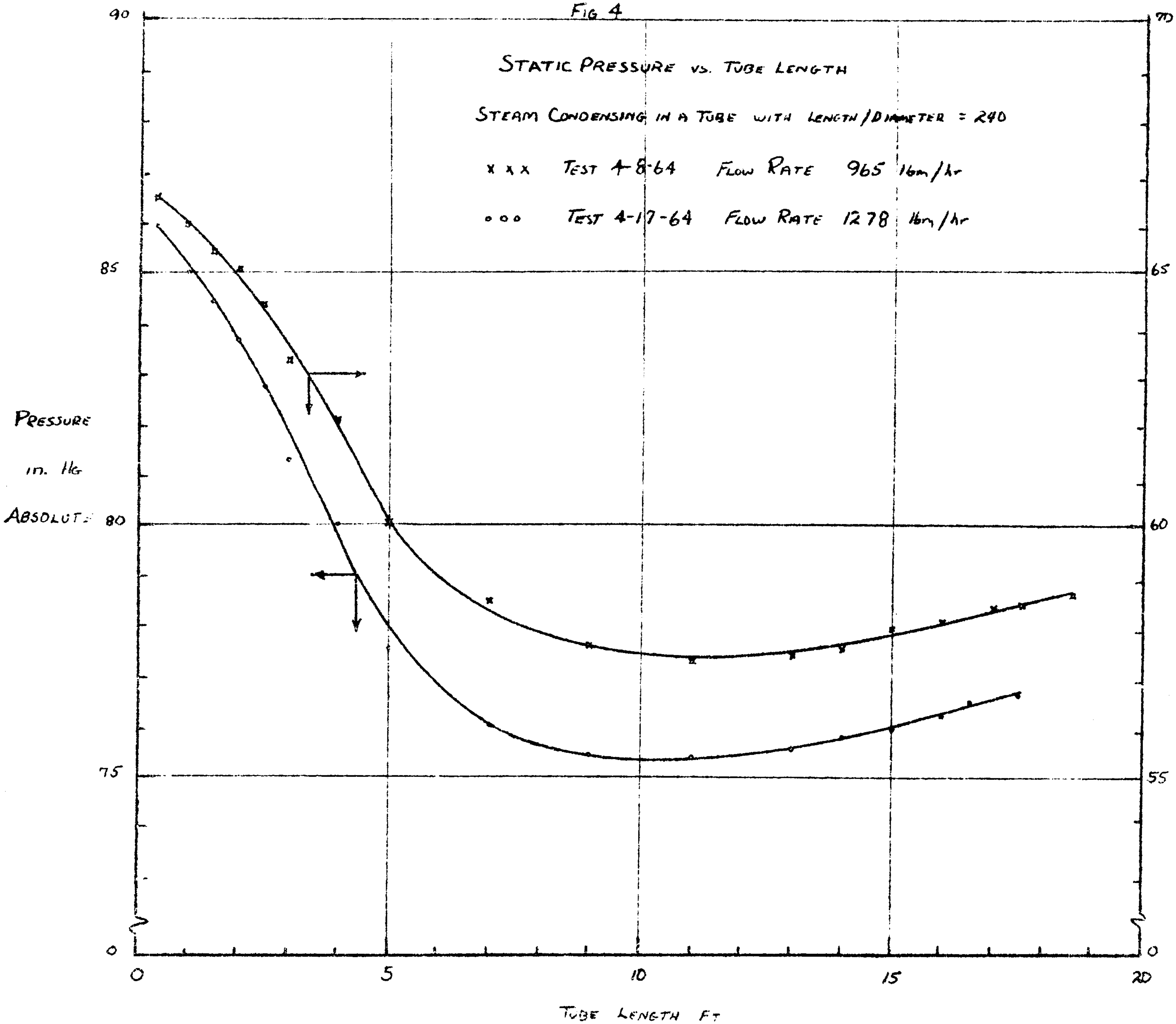
Another test section will be constructed utilizing glass in part of its length. Flow visualization studies will be made of the effect of probes inserted in the stream as well as studies of flow patterns under various pressure gradients.

The results of the experimentation discussed above will be used to gain a broader insight of the mechanisms of condensing flow. With this additional knowledge, if need be, a better mathematical model can be developed and a more meaningful solution, based upon physical principles, made of the equations.

Results

The greater part of the period covered in this report was spent in the design, construction and de-bugging of the test apparatus. A complete set of test data has not yet been obtained although several runs utilizing the velocity pressure probe have been made. These runs were made, however, in the interest of developing the test procedures rather than gathering data. Static pressure, tube wall and cooling water temperature data are available and Figs. 4 and 5 show two typical sets of these data.

Fig 4


Static Pressure vs. Tube Length
Steam Condensing in a Tube with Length/Diameter = 240
x x x Test 4-8-64 Flow Rate 965 lbm/hr
o o o Test 4-17-64 Flow Rate 1278 lbm/hr
Pressure in. Hg Absolute
90
85
80
75
0
70
65
60
55
0
5
10
15
20
Tube Length Ft

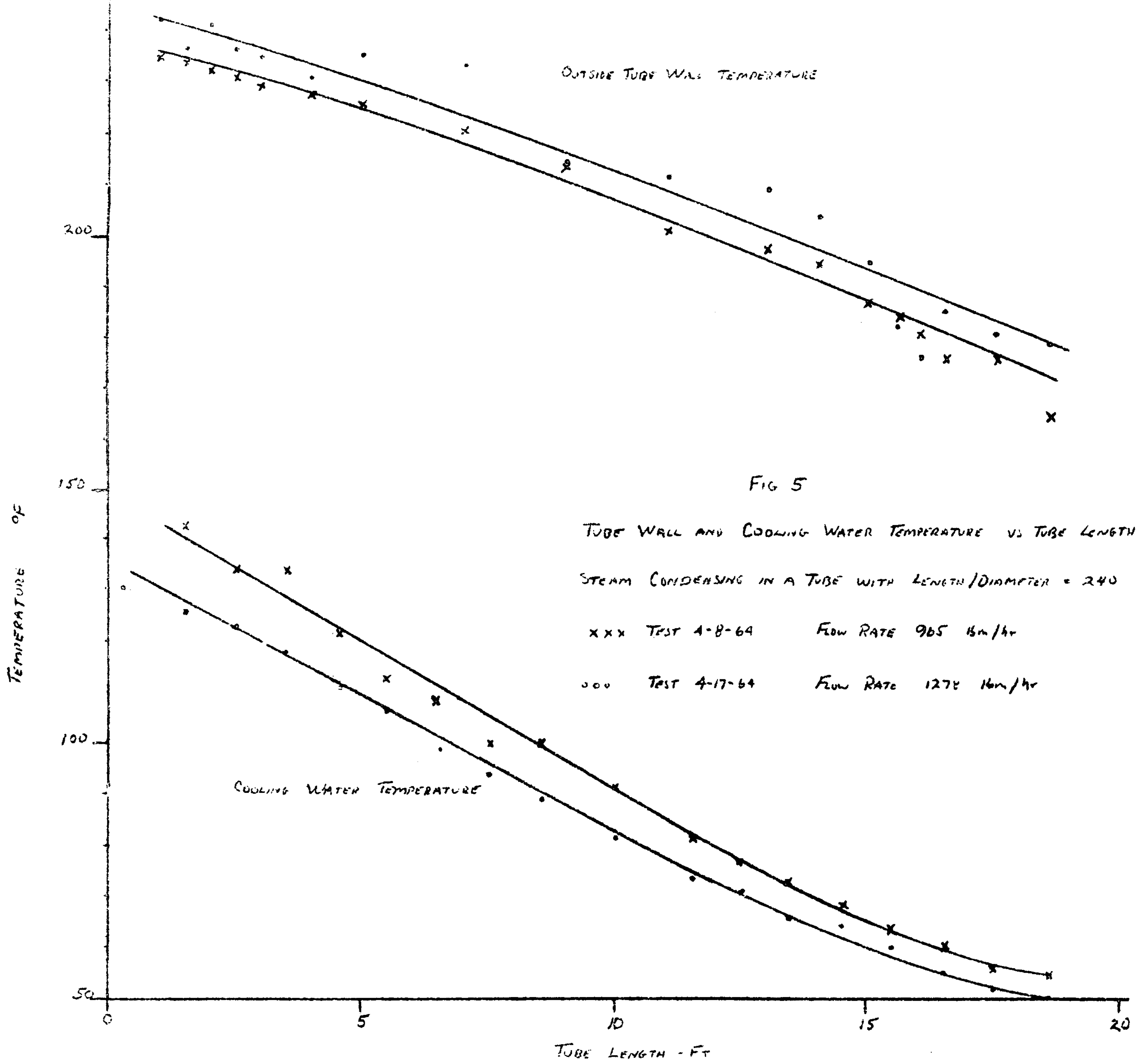


Fig 5

Tube Wall and Cooling Water Temperature vs Tube Length

Steam Condensing in a Tube with Length/Diameter = 240

Reference**s**

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6. SHARE #1095 "WH58, Thermodynamic Properties of Steam and Water."